NACA

RESEARCH MEMORANDUM

DESIGN AND TEST OF MIXED-FLOW IMPELLERS

VI - PERFORMANCE OF PARABOLIC-BLADED IMPELLER WITH SHROUD REDESIGNED BY RAPID APPROXIMATE METHOD

By Kenneth J. Smith and Walter M. Osborn

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NATIONAL ADVISORY COMMITTEE FOR AERONAUTICS

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SUMMARY

In an experimental investigation, a centrifugal impeller for which experimental and theoretical data were available was modified by a recently developed design procedure to reduce the velocity gradients existing along the hub from inlet to outlet. The performance of the modified impeller was investigated over a range of equivalent impeller tip speeds from 900 to 1500 feet per second and over a range of flow rates from maximum to the point of incipient surge. At a speed of 1300 feet per second, the peak pressure ratio and maximum adiabatic temperature-rise efficiency (based on measurements taken at a radius twice the impeller-outlet radius in a vaneless diffuser) were 3.07 and 0.825, respectively. At the maximum speed of 1500 feet per second, the peak pressure ratio was 4.03 and the maximum efficiency 0.800. The maximum efficiency points over the speed range tested were between 0.854 and 0.800. Dumping the air at the diffuser outlet into the test collector from a Mach number of approximately 0.45 resulted in losses of 0.09 to 0.14 in efficiency over the speed range tested.

The modified impeller had better performance characteristics than the original at all speeds tested, the greatest gains in performance occurring at speeds above 1300 feet per second. At 1500 feet per second, the efficiency of the modified impeller was 0.800 compared with 0.605 for the original impeller. The large gains in performance at higher speeds resulted largely from the decreasing tendency toward eddy formation that enabled the modified impeller to operate over a wider range of weight flows.

An arbitrary allowance for boundary-layer growth of approximately 30 percent of the design blade height at the impeller outlet and varying linearly to zero allowance at the impeller inlet is too small for the modified impeller with respect to static-pressure distribution along the shroud.

INTRODUCTION

In 1947, the NACA Lewis laboratory investigated the effects of blade curvature on centrifugal-impeller performance by using three impellers of identical geometry but with different blade shapes. The blade shapes tested were parabolic, elliptical, and circular. The performance results for the three impellers, reported in reference 1, indicated fair low-speed operating characteristics with maximum efficiencies between 0.75 and 0.81. However, performance results at equivalent speeds above 1200 feet per second showed poor efficiencies for the three impellers and unstable operating characteristics for the elliptical- and circular-bladed impellers.

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During 1949 and 1950, a method was developed at the Lewis laboratory for analyzing the compressible flow between the hub and shroud of mixed-flow impellers of arbitrary design. This method is presented in reference 2, along with the results obtained by applying the analytical method to the parabolic-bladed impeller of reference 1. The analysis showed a large variation in velocity across the passage from the hub to the shroud of the parabolic-bladed impeller and indicated that velocity gradients conducive to boundary-layer build-up and flow separation existed along both the hub and shroud profiles.

In 1950, the laboratory began a program of research on centrifugal compressors to evolve reliable design procedures that would eliminate a large amount of experimental development and increase the efficiency of the centrifugal compressor. Two mixed-flow impellers, the MFI-1 and MFI-2, of radically different geometric configurations, were designed by the method of reference 3 and analyzed by the method of reference 2. Theoretical and experimental results of the investigations of these two impellers are given in a series of reports (refs. 4 to 8). Both impellers had high efficiencies and good operating characteristics over the speed range tested (700 to 1700 ft/sec equivalent speed for MFI-1, and 700 to 1400 for MFI-2) and compared favorably with designs that required considerable developmental work (refs. 9 and 10, e.g.).

The design procedure used for the MFI-1 and MFI-2 impellers consisted in deriving the blade shape for a tentative hub-shroud shape by the method of reference 3 and then adjusting the hub-shroud shape to meet changes dictated by an analysis of the flow in the hub-shroud plane by the method of reference 2. Thus, the analytical method was utilized as a design tool. Since the analytical method requires a lengthy iteration process, the time required for such a solution is almost prohibitive. Therefore, a simplified design method using the same equations as in reference 2 was evolved from the analytical method. This design method, presented in reference 11, gives the average velocities between blades. The velocities on the blade surfaces may be obtained from these values by the method presented in reference 2. The resulting quasi-three-dimensional solution gives a picture of the velocity gradients on all surfaces.

Since considerable theoretical and experimental data were available for the parabolic-bladed impeller of reference 1, the impeller was redesigned by the method of reference 11 for further testing. The redesign of this impeller is given as the numerical example of reference 11.

This report presents the performance characteristics of the redesigned impeller and compares the performance with that of the original parabolic-bladed impeller of reference 1.

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SYMBOLS

The following symbols are used in this report:

f_s slip factor, ratio of absolute tangential velocity at exit to impeller speed at exit, approximated by ratio of measured enthalpy rise to U^2/gJ

g acceleration due to gravity, 32.2 ft/sec^2

J mechanical equivalent of heat, 778.2 ft-lb/Btu

L ratio of distance along impeller shroud from inlet to total length of shroud

r radius from axis of rotation, in.

U actual impeller tip speed, ft/sec

w actual air weight flow, lb/sec

x axial distance from hub at 6-inch radius (impeller outlet) measured positive toward inlet, in.

y peripheral distance along cylindrical surface of radius r, in.

z axial distance from entrance edge of blades, in.

δ ratio of inlet total pressure to NACA standard sea-level pressure of 29.92 in. Hg abs

η_{ad} adiabatic temperature-rise efficiency

θ ratio of inlet total temperature to NACA standard sea-level temperature of 518.7° R

MODIFIED IMPELLER DESIGN

A comparison of the analytical data for the MFI-1 impeller (ref. 4), the MFI-2 impeller (ref. 6), and the parabolic-bladed impeller (ref. 2) showed the following results: the MFI-1 impeller (the most efficient) had accelerating velocities along the hub profile; the MFI-2 impeller had accelerating velocities along most of the hub profile but had decelerating velocities near the inlet and outlet sections; the original parabolic-bladed impeller (least efficient) had decelerating velocities (steep velocity gradients) along most of the hub.

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Perhaps if accelerating flow could be accomplished along the hub profile of the parabolic-bladed impeller, the efficiency would be increased. Therefore, accelerating velocities were assigned along the hub profile of the parabolic-bladed impeller and a new shroud contour was obtained by the design method of reference 11. A preliminary investigation indicated that the Mach number along the new shroud would be excessive, so the assigned velocities were changed to slightly decelerating velocities from the inlet to the midportion of the hub profile and slightly accelerating velocities from the midportion to the outlet of the hub profile. These changes resulted in a shroud contour on which the velocities were no higher than the inlet velocity. An arbitrary allowance for boundary layer was made because the resulting shroud profile is for isentropic flow. This allowance consisted of 30 percent of the design blade height at the impeller outlet and varied linearly from the outlet to zero allowance at the impeller inlet.

The shroud profiles for the original impeller, the design impeller (isentropic flow), and the modified impeller (design impeller plus arbitrary allowance for boundary layer) are compared in figure 1. The redesign of this impeller took approximately 45 hours of computing time for a 3-streamtube solution. The complete design procedure is given in the numerical example of reference 11.

The hub contour, blade shape, and blade taper of the parabolic-bladed impeller were such that a shroud contour of increasing radius from inlet to outlet resulted when slightly decelerating velocities from the inlet to the midportion of the hub profile and slightly accelerating velocities from the midportion to the outlet were assigned along the hub contour. With impellers of different geometric parameters, assigning accelerating velocities along the hub may result in a shroud-contour radius downstream of the impeller inlet that is smaller than the inlet radius. This would require a complicated assembly problem unless the impeller were fully shrouded. For an original design, this problem may be alleviated by altering the hub contour instead of changing the velocity profile along the hub.

APPARATUS, INSTRUMENTATION, AND PROCEDURE

Apparatus

The original parabolic-bladed impeller of reference 1, with a new shroud contour, was used for this investigation. This impeller has radial blade elements and 18 blades. The blade curvature, determined by the equation $y = 0.05655rx^2$, corresponds to that of a parabola on the developed surface of a cylinder and extends the full depth of the impeller. The coordinates for the modified shroud and the necessary information for building the impeller are given in table I.

The impeller was tested with a 25-inch-diameter vaneless diffuser of a constant area in the radial direction. There was 0.042-inch clearance (normal to impeller shroud) between the impeller shroud and the shroud wall with the impeller in a stationary position. A schematic diagram and a photograph of the impeller and diffuser are shown in figures 2 and 3, respectively. Adapters were made for the front and rear of the impeller to fit it to the existing shafting, the impeller being straddle-mounted between two bearings. This differs from the original installation (ref. 1), in which the impeller was supported by a rear bearing only (overhung). Thus, in the installation of reference 1, the radius of the outer wall was constant upstream of the inlet and converged at the inner radius (spinner); whereas, for the present installation the radius of the inner wall was constant and converged at the outer radius as shown in figure 2. The effect of this difference of inlet geometry on the performance of the impeller is unknown.

The remainder of the experimental setup is the same as that described in reference 5.

Instrumentation

The instrumentation is similar to that described in reference 5. The outlet measuring station is located at a 12-inch radius (twice the impeller-outlet radius) in the vaneless diffuser as shown in figure 2 and is at the same radius as in the original installation of reference 1. The diffuser instrumentation consisted of eight static-pressure taps, four thermocouple rakes, and 12 total-pressure probes. Four static-pressure taps were located in the front diffuser wall at 90° intervals around the annulus opposite four static-pressure taps in the rear diffuser wall. The four thermocouple rakes were also placed 90° apart and had three thermocouples per rake spaced at intervals of 1/6, 1/2, and 5/6 of the distance across the passage. The 12 total-pressure probes were distributed around the annulus to give a coverage for total pressure equivalent to that of the temperature measurements. In addition, 11 static taps were located along the shroud wall from the impeller inlet to the outlet.

In order to evaluate the over-all efficiency and pressure ratio, total-temperature and total-pressure measurements were taken in the outlet pipe (7 impeller diameters downstream). Temperature measurements taken in the outlet pipe agreed with the temperature measurements taken in the diffuser within the limits of experimental error.

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Procedure

This investigation was carried out at a constant inlet-air pressure of 20 inches of mercury absolute. The inlet temperatures varied from ambient to -55° F. It was necessary to run at 900 feet per second with -45° F inlet-air temperature in order to avoid the drive-motor critical-speed range. The flow rate was varied from maximum to the point of incipient surge by varying the outlet pressure.

The impeller equivalent speed was varied from 900 to 1500 feet per second based on an impeller-outlet radius of 6 inches. Data for an equivalent speed of 1600 feet per second could not be obtained because of the speed limitation of the rig.

The test and computational procedures are the same as those used in reference 5.

EXPERIMENTAL RESULTS

The over-all performance characteristics for the modified parabolic-bladed impeller with a vaneless diffuser are presented in figure 4 for a range of speed from 900 to 1500 feet per second. The peak pressure ratio and maximum adiabatic efficiency at 1300-feet-per-second equivalent speed were 3.07 and 0.825, respectively. At the maximum speed of 1500 feet per second, the peak pressure ratio was 4.03 and the maximum efficiency was 0.800. The average Mach number at the outlet measuring station for the maximum efficiency points over the range of speed was between 0.36 and 0.44 (fig. 4(b)).

Experimental results for the modified and original impellers are compared in figure 5. The modified impeller had a higher peak total-pressure ratio than the original impeller at all speeds investigated (fig. 5(a)). The greatest gain in peak pressure ratio for the modified impeller was at 1500-feet-per-second equivalent tip speed, where the pressure ratio was 4.03 compared with 3.21 for the original impeller. The modified impeller had maximum efficiencies between 0.854 and 0.800 over the speed range tested (fig. 5(b)). These efficiencies were higher than those of the original impeller at all speeds tested, with a gain in efficiency of 0.195 (0.605 to 0.800) at 1500-feet-per-second equivalent speed. The slip factors of the original impeller were higher than those of the modified impeller at all speeds (fig. 5(c)).

The performance of the modified impeller as determined by measurements taken at two times the impeller-outlet radius in the diffuser (fig. 2) and the performance determined by measurements taken in the outlet pipe (over-all performance) are compared in figure 6. Efficiencies based on measurements taken in the outlet pipe were approximately 0.09 less than those in the diffuser between speeds of 1200 and 1500 feet per second. The losses in efficiency increase at speeds below 1200 feet per second to approximately 0.140 at 900 feet per second. A loss
3626 in pressure ratio also results from basing the pressure ratio on total-pressure measurements in the outlet pipe.

The theoretical static-pressure ratio (static pressure/inlet total pressure) along the shroud from the impeller inlet to outlet at the design point ($U\sqrt{\theta}$ = 1331 ft/sec and $w\sqrt{\theta}/\delta$ = 8.38 lb/sec) and the experimental static- to inlet-total-pressure ratios at weight flows of 8.49, 8.05, and 7.53 pounds per second are compared in figure 7. The general trends of experimental static-pressure ratios from impeller inlet to outlet are similar to the theoretical static-pressure ratios.

DISCUSSION OF RESULTS

The performance of the modified parabolic-bladed impeller with vaneless diffuser showed considerable improvement over that of the original impeller (fig. 5) and compares favorably with the performance level of the MFI-2 impeller (ref. 7). The performances of the MFI-1 and -2 impellers were based on measurements taken at a radius $1\frac{1}{2}$ times the impeller-outlet meanline radius in a vaneless diffuser where the Mach number was approximately 0.65. The Mach number for the modified impeller at the diffuser measuring station (twice the impeller-outlet radius) was 0.45. Thus, the efficiency of the modified impeller probably falls between the efficiencies of the MFI-1 and -2 impellers. The lower efficiency of the modified impeller as compared with the efficiency of the MFI-1 impeller may result in part from the less evenly distributed blade loading (refs. 4 and 11) of the modified impeller.

The increase in performance of the modified impeller over that of the original impeller may be explained by a study of the internal-flow characteristics of each. An analysis of the flow in the blade-to-blade plane of the original impeller (1331-ft/sec impeller equivalent speed) showed a large eddy on the driving face of the blade at the hub. Eddy formation is taken herein as beginning when the theoretical velocity on the blade surface becomes negative. If, experimentally, the reversal of flow that accompanies the formation of the eddy causes instability, a separation and rotating stall such as discussed in reference 12 may occur. If the rotating stall does not result in surge or results in surge mild enough to allow operation at a lower weight flow, a second stall or surge point caused by too large a relative inlet angle will be reached. There are two surge points for some operating speeds shown in figure 4 of reference 12 that may be the result of such an occurrence. For the original parabolic-bladed impeller, the unstable eddy apparently caused violent surge at speeds of 1400 and 1500 feet per second (fig. 5(a)), whereas at lower speeds it merely caused a reduction in efficiency with decreasing weight flow.

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In redesigning the original impeller, the tendency toward eddy formation was decreased by increasing the theoretical velocity ratio (ratio of velocity relative to impeller to stagnation speed of sound upstream of impeller inlet) from 0.17 to 0.56 in the region where the eddy occurs. Thus, the increased performance of the modified impeller over the original impeller (fig. 5) results from a combination of the reduced velocity gradients and the decrease of the tendency toward eddy formation, which enabled the modified impeller to operate successfully at high speeds. However, the shift of the surge line of the modified impeller to lower weight flows than for the surge line of the original impeller at speeds below 1200 feet per second may be due to the difference in the inlet geometry upstream of the impellers.

The difference in pressure ratio of the modified and original impellers (fig. 5(a)), is small at the 900- and 1100-foot-per-second speeds, but the difference in efficiency (fig. 5(b)) is large (0.05 to 0.07). This may be accounted for by the poor internal-flow characteristics of the original impeller, which cause particles of air inside the impeller that have had whirl imparted to them to flow upstream (backflow) into the inlet section ahead of the impeller. These particles of air are then forced to the inlet shroud by centrifugal force and reenter the impeller at a higher temperature than if there had been no backflow. Thus, the large difference in efficiencies between the two impellers at low speeds may be attributed in part to the low efficiency of the original impeller caused by an increased outlet temperature due to backflows at the impeller inlet. This backflow phenomenon also contributes to the dropping off of the efficiency of the modified impeller at the 900-foot-per-second speed at weight flows less than 4.75 pounds per second, with the point at 2.05 pounds per second probably in mild surge (inaudible). In actual violent surge (audible), the backflow phenomenon could be detected by an increase in the inlet temperature in the surge tank approximately 7 impeller diameters upstream of the impeller inlet. A study of backflow phenomenon is presented in reference 13.

The slip factor for the original impeller is only slightly higher than that of the modified impeller at speeds between 900 and 1200 feet per second, but at higher speeds the difference becomes large (fig. 5(c)). Since the slip factor is approximated by the over-all enthalpy rise, an increased outlet temperature due to backflow in the original impeller would cause the slip factor to be too high. Thus, the slip factors for the modified impeller are probably more accurate than those for the original impeller. The large difference in slip factors of the two impellers above 1200 feet per second probably results from backflow effects in the original impeller ensuing from a stronger tendency for eddy formation at the higher speeds. Reference 14 shows theoretically that, for impellers with radial blades at the exit, the slip factor depends solely upon the number of blades and thus should be constant over the speed range. The modified impeller more nearly approaches a constant slip factor than does the original impeller (fig. 5(c)).

The comparison of modified-impeller performances (fig. 6) based on measurements taken in the diffuser and in the outlet pipe shows a drop in efficiency of 0.09 to 0.14 over the speed range tested by dumping the air at the end of the diffuser into the test collector at a Mach number of approximately 0.45. The loss in pressure ratio is 8 to 12 percent. Even though the losses associated with dumping the air were large, the over-all efficiency at a speed of 1300 feet per second (theoretical design speed is 1331 ft/sec) was approximately 0.74.

In actual practice, the air would not be dumped from a Mach number of 0.45 into a collector such as used for these tests. Instead, a vaneless diffuser of larger diameter, a scroll-type diffuser, diffuser vanes, or a combination of these might be used to further reduce the Mach number before dumping. For example, an efficient set of diffuser vanes might be placed after the existing vaneless diffuser to reduce the Mach number from approximately 0.45 to 0.20. Reference 15 presents the test results for two sets of diffuser vanes and indicates a loss of only 0.017 to 0.037 in efficiency over the speed range tested when the air is dumped from a Mach number of approximately 0.20. In reference 16, the performance of diffusers is evaluated in terms of total-pressure-loss coefficient (ratio of mean total-pressure loss to mean dynamic pressure at inlet). The curves of reference 16 give a loss coefficient of 0.22 at a Mach number of 0.45 for a diffuser with two parallel walls and two walls diverging at an included angle of 10.6°. By taking the data for the modified impeller at a speed of 1500 feet per second and placing a set of diffuser vanes having a loss coefficient of 0.22 behind the existing diffuser an efficiency of 0.782 would be possible at a Mach number of approximately 0.20 at the vane exit. The dumping loss in efficiency associated with a speed of 1500 feet per second and a Mach number of 0.20 is given in reference 15 as approximately 0.018. Thus, for the modified impeller at a speed of 1500 feet per second, it might be possible to obtain an over-all efficiency (based on measurements taken in the outlet pipe) of approximately 0.764. This would give a gain in over-all efficiency of approximately 0.050 over the efficiency without diffuser vanes.

The comparison of theoretical and experimental static-pressure ratios along the impeller shroud (fig. 7) may be used to indicate the accuracy of the arbitrary allowance for boundary-layer growth made during the design procedure for the modified impeller. The arbitrary boundary-layer allowance was 30 percent of the design blade height at the impeller outlet and varied linearly from the outlet to zero allowance at the impeller inlet. This allowance was chosen on the basis of the experimental results obtained with the MFI-1 and -2 impellers (ref. 7). The general trends of the experimental and theoretical static-pressure ratios from the impeller inlet to outlet are similar. However, the static-pressure ratios for an experimental weight flow of 8.49 pounds per second (theoretical weight flow is 8.38 lb/sec) are lower than the theoretical static-pressure ratios, especially near the impeller outlet. At an experimental weight flow of 7.53 pounds per second, the static-pressure ratios agreed with the theoretical pressure ratios near the impeller outlet. In order to have the same experimental and theoretical static-pressure ratios at the impeller outlet (weight flow is 8.38 lb/sec), it would be necessary to have a boundary-layer allowance of approximately 45 percent of the design blade height at the impeller outlet. Thus, on the basis of static-pressure ratios along the impeller shroud, the boundary-layer allowance for this impeller was too small.

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Reference 15 shows that the experimental velocity distribution varies greatly from the theoretical velocity distribution at the outlet of the MFI-1 impeller because of boundary-layer build-up along the shroud. Therefore, it would be difficult to determine the optimum boundary-layer allowance from a consideration of the theoretical velocity distribution. Instead, an experimental technique such as was used with the MFI-1 and -2 impellers may be required, in which varying allowances for boundary layer were made in an attempt to find the one which allowed maximum efficiency.

SUMMARY OF RESULTS

A centrifugal impeller with good performance characteristics was obtained by assigning a new velocity distribution that reduced the velocity gradients from inlet to outlet along the hub of an existing centrifugal impeller of mediocre performance and applying the design procedure of reference 11 to arrive at a new shroud contour. An investigation of the performance characteristics of this modified centrifugal impeller (based on measurements taken at a radius twice the impeller radius in a vaneless diffuser and 7 diameters downstream in the outlet pipe) produced the following results:

1. The peak pressure ratio and maximum adiabatic efficiency at 1300-feet-per-second equivalent speed were 3.07 and 0.825, respectively. At a maximum speed of 1500 feet per second, the peak pressure ratio was

4.03 and the maximum efficiency was 0.800. The maximum efficiencies over the speed range tested were between 0.800 and 0.854.

2. The modified impeller, as compared with the original impeller, had higher peak pressure ratios and maximum efficiencies at all speeds investigated. The efficiency at a speed of 1500 feet per second was 0.800 compared with 0.605 for the original impeller.

3. The large gain in performance of the modified impeller compared with that of the original impeller at speeds above 1300 feet per second probably resulted largely from the decrease of the tendency toward eddy formation. The modified impeller was thus able to operate over a wider range of weight flows at high speeds.

4. Over the speed range tested, losses in efficiency of 0.09 to 0.14 result from dumping the air at the end of the diffuser into the test collector at a Mach number of approximately 0.45.

5. An arbitrary allowance for boundary-layer growth amounting to 30 percent of the design blade height at the impeller outlet and varying linearly from the outlet to zero allowance at the impeller inlet is too small for this impeller with respect to static-pressure distribution along the shroud.

Lewis Flight Propulsion Laboratory
National Advisory Committee for Aeronautics
Cleveland, Ohio, July 15, 1955

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TABLE I. - GEOMETRIC COORDINATES OF MODIFIED PARABOLIC-BLADED IMPELLER

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Axial depth, z, in. (fig. 2)	Modified shroud radius, r, in. (fig. 2)	Axial depth, z, in. (fig. 2)	Modified shroud radius, r, in. (fig. 2)
0 (leading edge)	4.000	2.884	5.250
.754	4.125	2.964	5.375
1.442	4.250	3.034	5.500
1.823	4.375	3.095	5.625
2.087	4.500	3.151	5.750
2.442	4.750	3.200	5.875
2.694	5.000	3.244	6.000

Hub coordinates given in ref. 1.

Material, aluminum.

Number of blades, 18.

Blade shape determined by equation $y = 0.05655rx^2$.

Constant blade thickness along hub contour of 0.210 in. with 3° included-angle blade taper symmetrical about radial line through hub determining thickness at other points. Leading edge thinned and rounded off.

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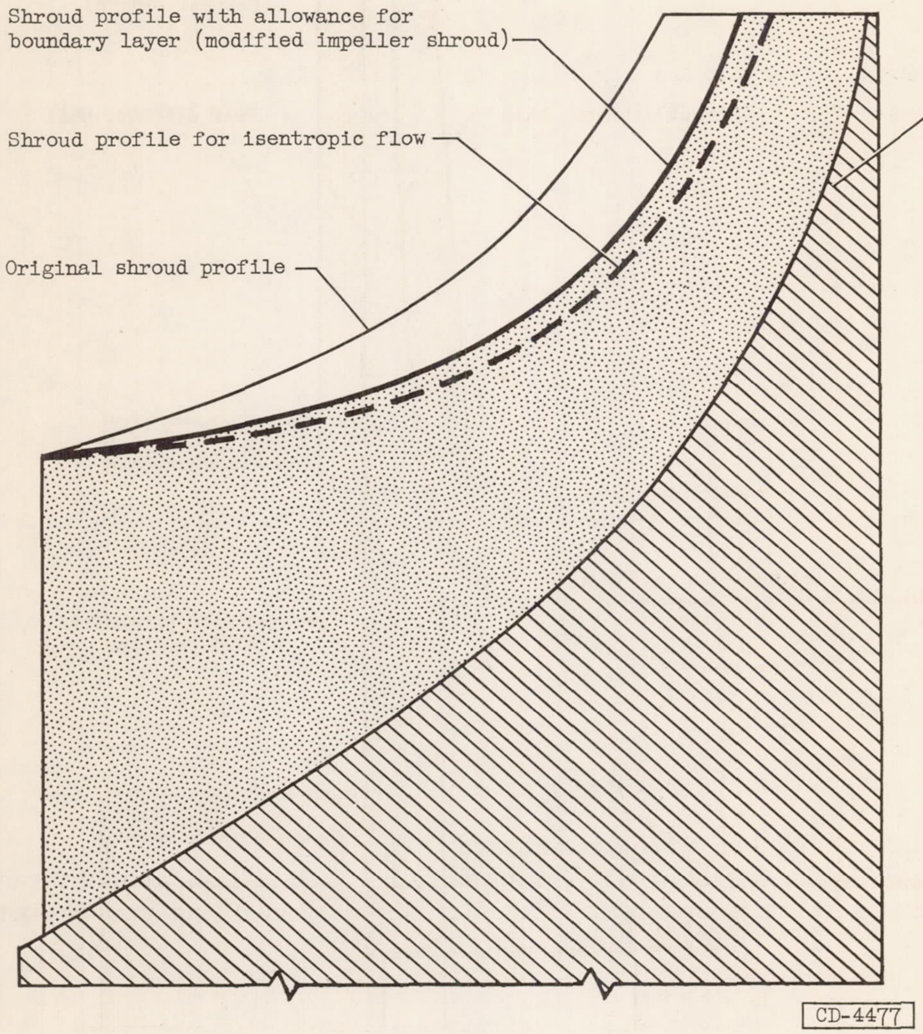

Figure 1. - Comparison of shroud profiles for original, design (isentropic flow), and modified impeller.

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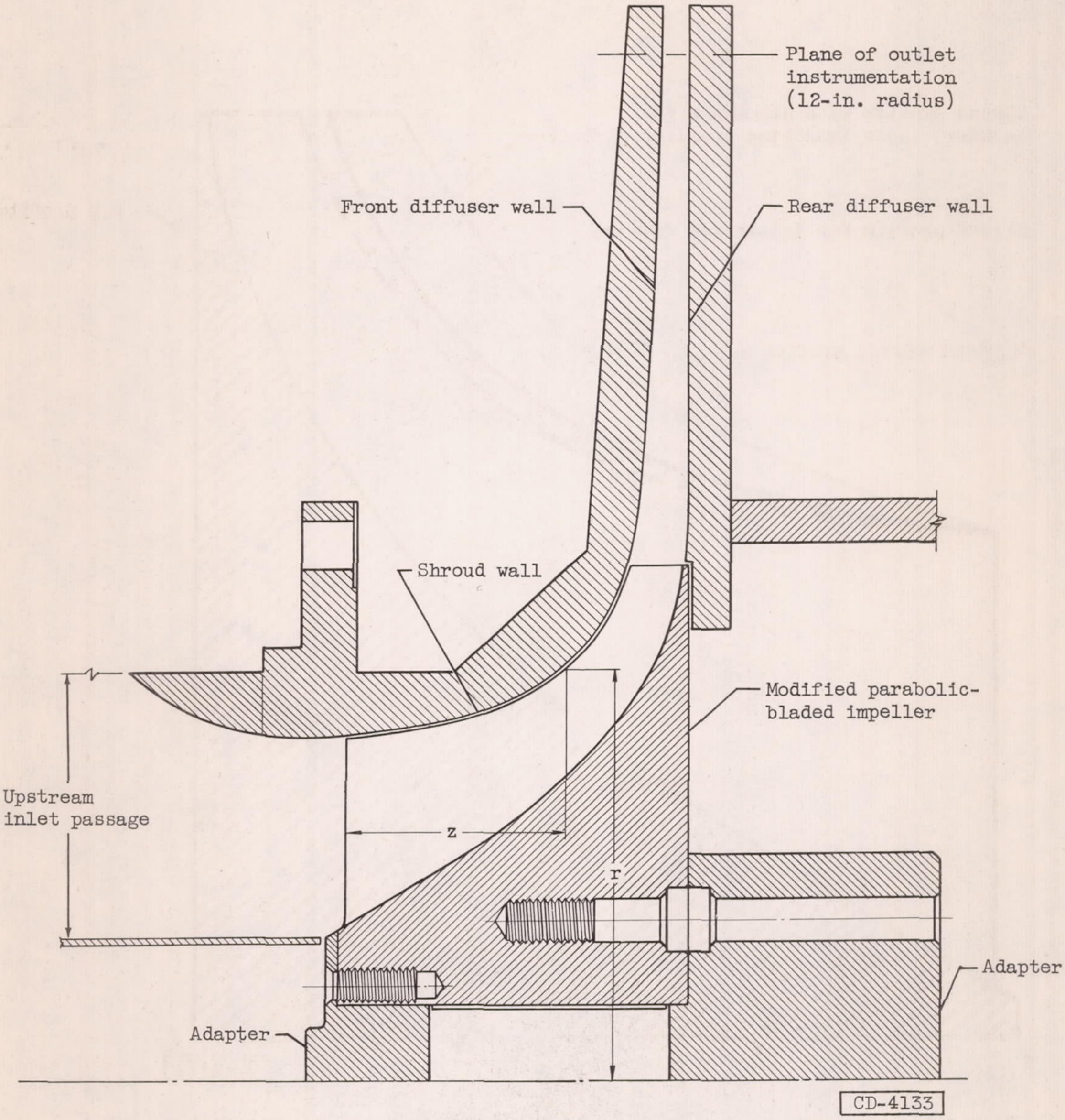

Figure 2. - Cross-sectional view of modified parabolic-bladed impeller and diffuser showing location of outlet instrumentation.

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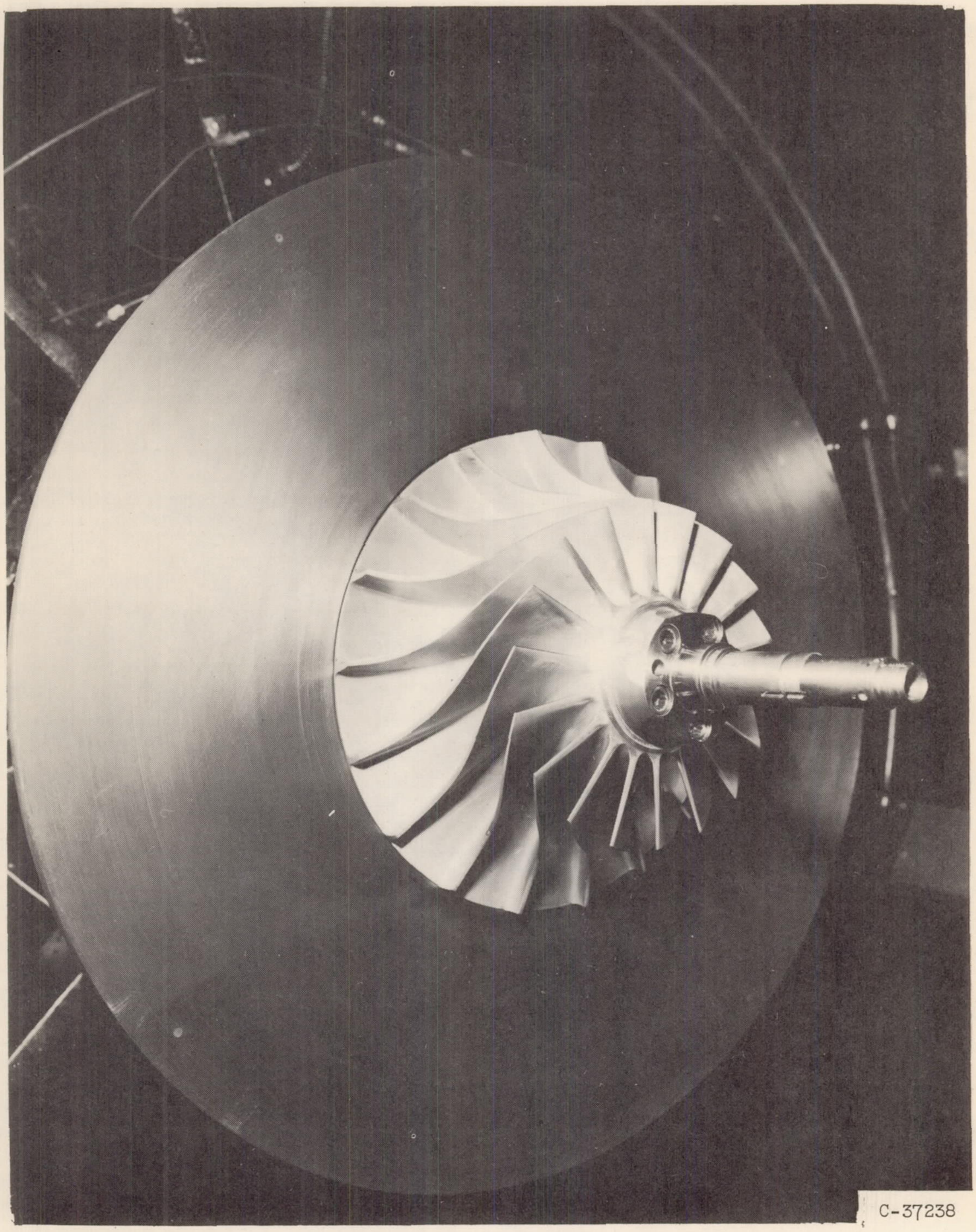

Figure 3. - Modified parabolic-bladed impeller and rear diffuser wall.

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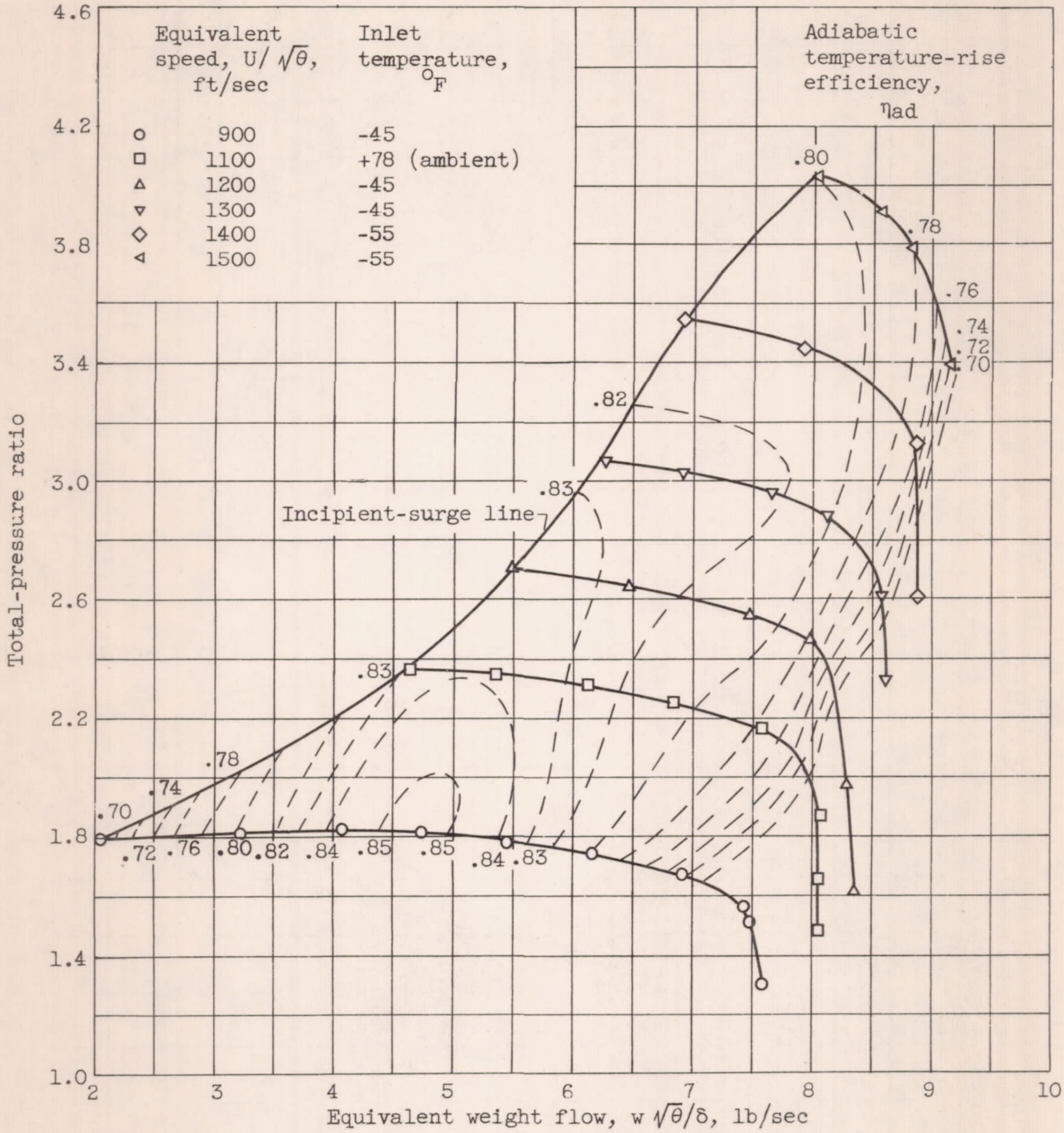
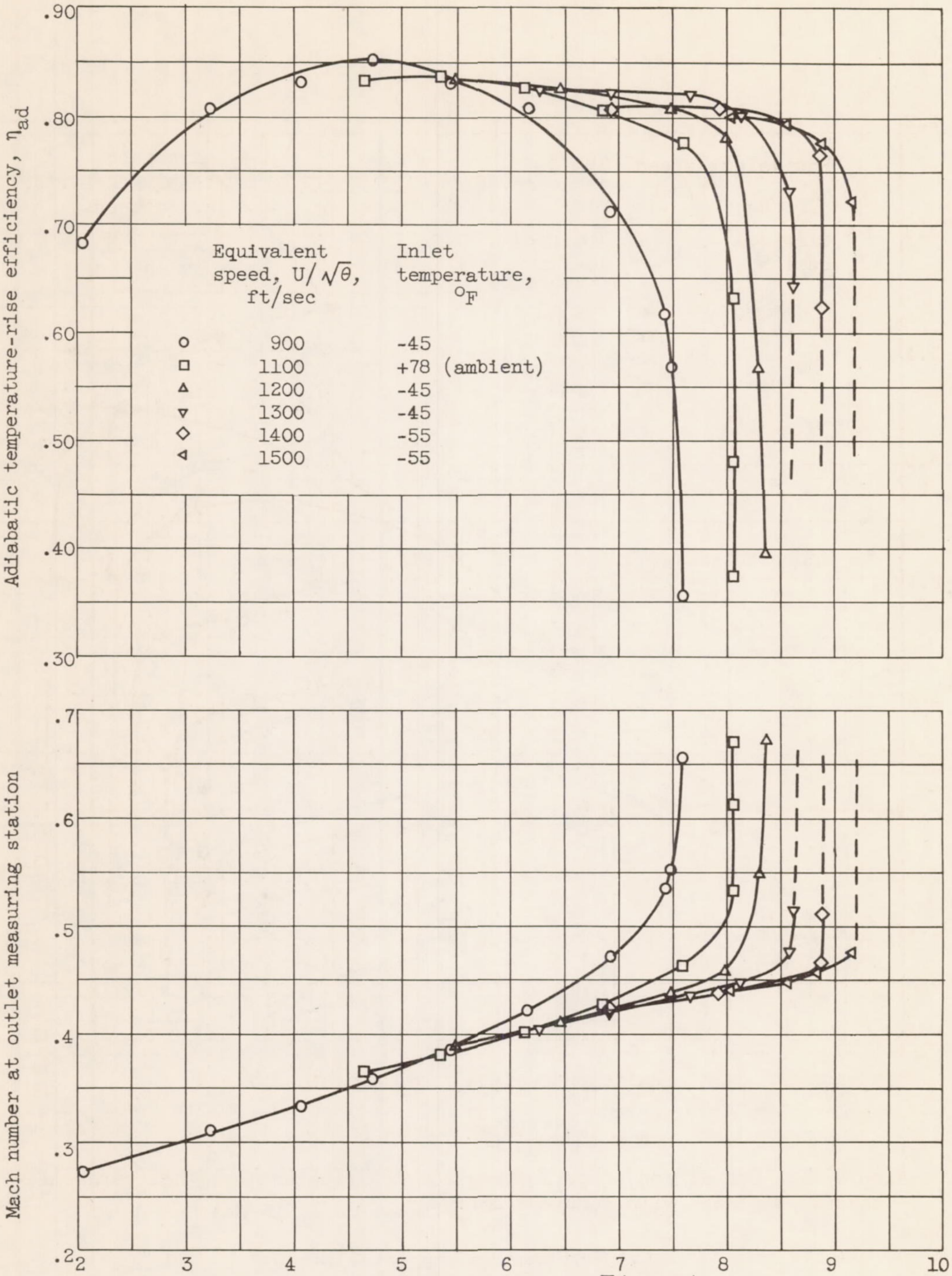


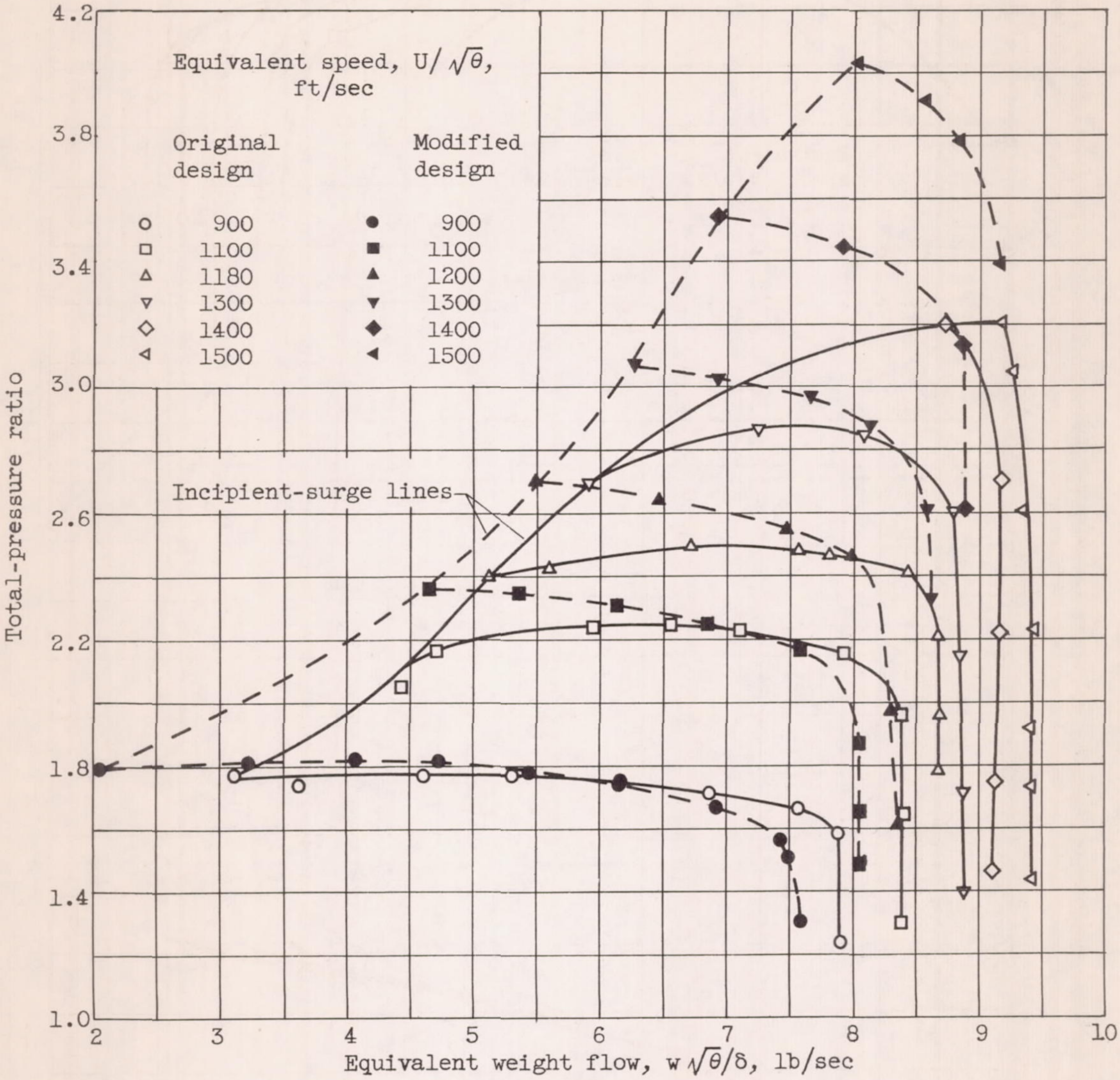
(a) Performance characteristics.

Figure 4. - Over-all performance characteristics of modified parabolic-bladed impeller with vaneless diffuser at inlet-air pressure of 20 inches of mercury absolute.



(b) Over-all efficiency and Mach number.

Figure 4. - Concluded. Over-all performance characteristics of modified parabolic-bladed impeller with vaneless diffuser at inlet-air pressure of 20 inches of mercury absolute.



(a) Total-pressure ratio.

Figure 5. - Comparison of performance of modified and original parabolic-bladed impellers with vaneless diffuser.

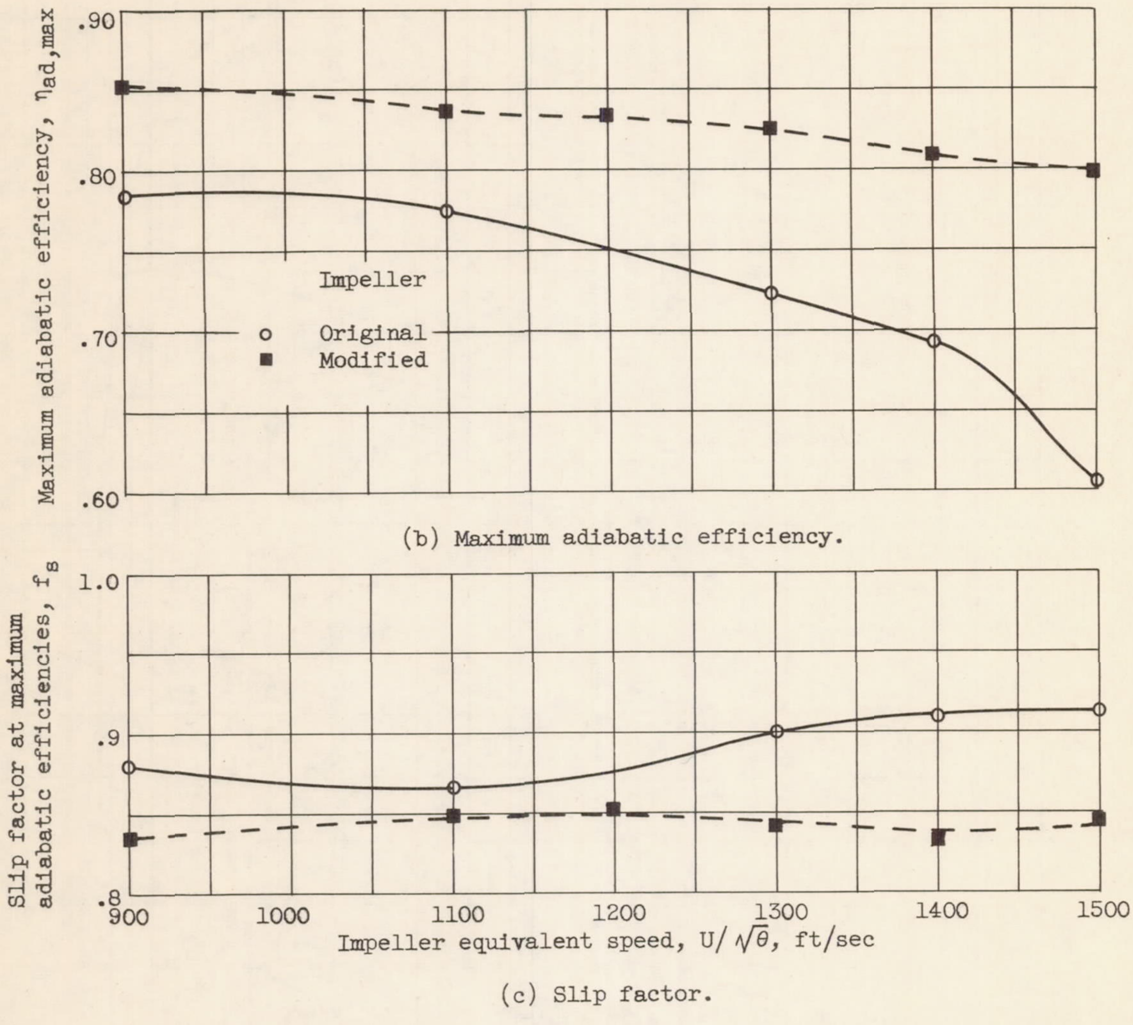


Figure 5. - Concluded. Comparison of performance of modified and original parabolic-bladed impellers with vaneless diffuser.

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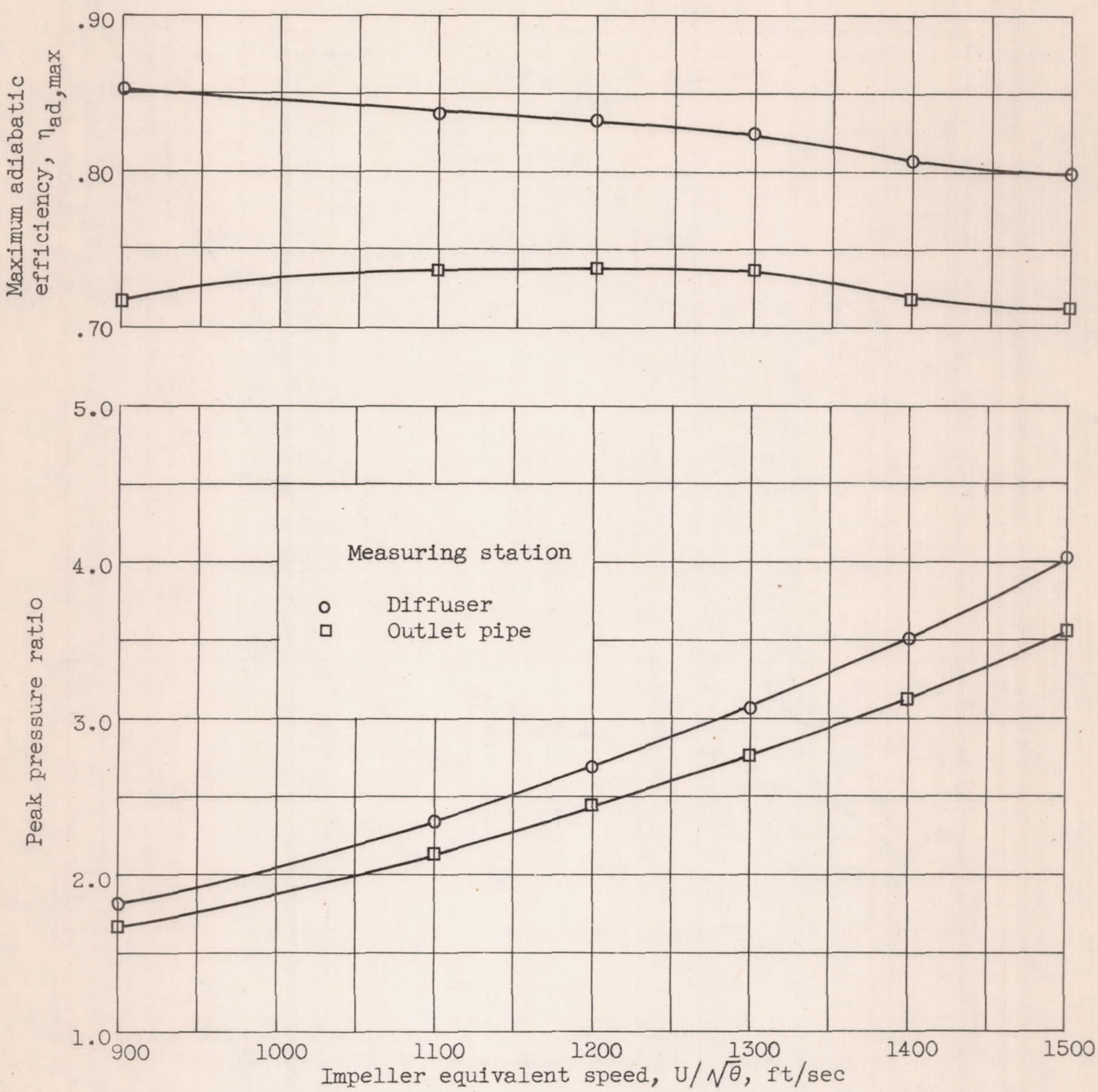

Figure 6. - Comparison of performance of modified parabolic-bladed impeller with vaneless diffuser based on measurements taken in diffuser (twice impeller-outlet radius) and in outlet pipe (7 impeller diameters downstream).

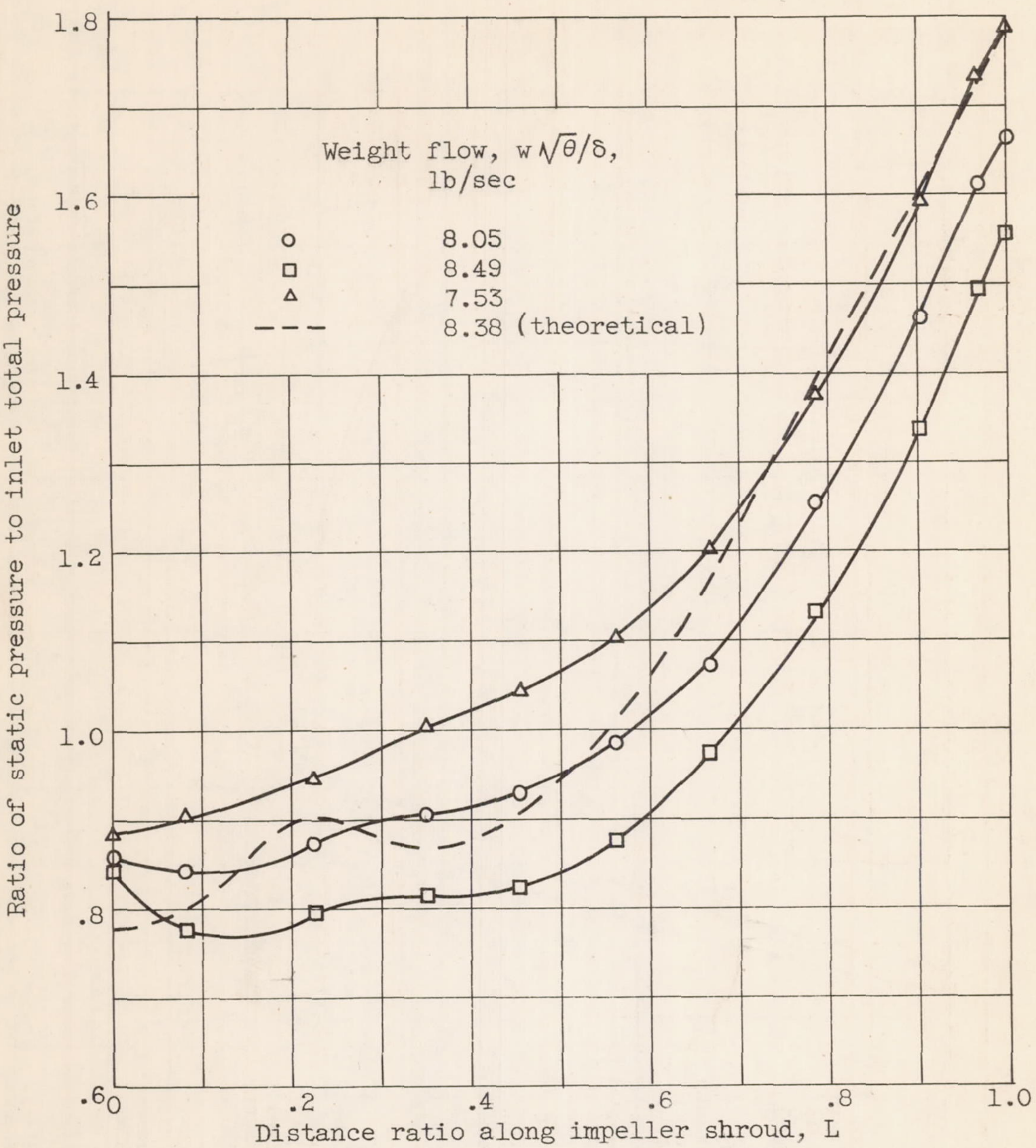

Figure 7. - Comparison of theoretical and experimental static pressures along shroud for modified parabolic-bladed impeller at design point. Impeller tip speed, 1331 feet per second; air weight flow, 8.38 pounds per second.

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